MODELLING OF A MUFFLE FURNACE USED IN FIROZABAD GLASS INDUSTRY

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by

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26/2/98

CERTIFICATE

This is to certify that the work contained in the thesis entitled "Modelling of a muffle furnace used in Firozabad Glass Industry" by Renu Thakur, has been carried out under my supervision and that this work has not been submitted elsewhere for a degree.

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ABSTRACT

An energy audit carried out by the Tata Energy Research Institute shows muffle furnace used in the Firozabad glass industry operate at very low efficiencies, of the order of 2%. Considerable savings in fuel costs would be achieved if the energy efficiency of these muffle furnaces is increased even moderately. Hence the objective of this research has been to develop a mathematical model which can be used for the energy-efficient design of the Firozabad muffle furnaces. The mathematical model involves the simultaneous solution of Bernoulli equation and energy balance equations. Bernoulli's equation has been used to determine the mass flowrate of gas inside the muffle furnace by natural draught. Computations showed that the calculated hearth and flue gas tempearture did not match well with measurements. The discrepancy between predicted and measured temperatures can be attributed to the low mass flowrate of flue gas within the furnace which implies that the assumption of constant hearth temperature in our model may not be appropriate.

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CONTENTS

NOMENCLATURE

$\dot{m}$	=	mass flow rate
C_p	=	specific heat
C_f	=	correction factor
V	=	volume of coal bed
h_v	=	convective heat transfer coefficient across the vertical walls.
k	=	thermal conductivity
T_{coal}	=	temperature of coal bed
T_a	=	ambient temperature
T_{h1}	=	temperature inside the bottom hearth
T_{h2}	=	temperature inside the middle hearth
T_{h3}	=	temperature inside the topmost hearth
T_0, T_{N1}	=	temperature at the extremeties of flue gas stream 0,1
T_{N1}, T_{N2}	=	temperature at the extremeties of flue gas stream 1,2
T_{N2}, T_{N3}	=	temperature at the extremeties of flue gas stream 2,3
T_{N3}, T_{N4}	=	temperature at the extremeties of flue gas stream 3,4
T_{N4}, T_{N5}	=	temperature at the extremeties of flue gas stream 4,5
T_{N5}, T_{N6}	=	temperature at the extremeties of flue gas stream 5,6
T_{N6}, T_{N7}	=	temperature at the extremeties of flue gas stream 6,7
T_{N7}, T_{N8}	=	temperature at the extremeties of flue gas stream 7,8
A	=	cross-sectional area
Q	=	Heat lost or heat gained
D	=	Thickness
NX	=	Number of divisions in the horizontal stream (0-1, 2-3, 4-5, 6-7)
NY	=	Number of divisions in the vertical stream (1-2, 3-4, 5-6)
NC	=	Number of divisions in the vertical stream flowing through the chimney

SUBSCRIPTS

01	=	related to flue gas stream 0,1
12	=	related to flue gas stream 1,2
23	=	related to flue gas stream 2,3
34	=	related to flue gas stream 3,4
45	=	related to flue gas stream 4,5
56	=	related to flue gas stream 5,6
78	=	related to flue gas stream 7,8
chm	=	related to chimney
coal	=	related to coal
h1	=	related to bottommost hearth
h2	=	related to middle hearth
h3	=	related to topmost hearth

1. *INTRODUCTION*

A furnace is a device in which the chemical energy or the electrical energy is converted into heat which is used to raise the temperature of materials.

The function of an industrial furnace is to generate and apply heat in such a manner that the heated product will conform to certain specifications, with lowest cost of heating per unit of finished products.

The selection of that source of heat which is best suited to a particular case necessitates the possession of the knowledge of - important properties of fuel, the equipments needed for their preparation the furnace for burning fuel economically with the correct amount of air, which varies with the nature of fuel, the devices for controlling furnace temperature and furnace atmosphere,cost of a given amount of heat when produced by different fuels.

The "economy" or "efficiency" when used in their true sense in connection with industrial furnaces, have reference to heating cost per unit weight of finished saleable product. "Heating cost" includes not only the cost of fuel but also the cost of firing the furnace, the cost of maintainence and repair, the cost of generating a protective atmosphere and cost of burnt, spoilt or rejected pieces.

With so many different items entering into the cost of heating,it is quite possible that in some cases, the highest priced fuel may be the cheapest in end, so far the total cost is concerned. With favourable conditions, excellent design, and with good operation appreciable furnace efficiencies can be obtained. By efficiency, it is meant the fuel efficiency which is the ratio of heat input into the stock to the potential heat of the fuel. Gases, for example, can give up heat to the charge only as long as they

are hotter than the charge. Consequently, the flue gases leave industrial furnace at a very high temperature. This factor is responsible for low furnace efficiency.

It is this requirement of low heat cost per unit of heated products that has led to the development of concept of the energy audits. These audits take into account the various factors, (discussed in background) leading to heat loss and hence affecting the economy of the process adversely. One such audit carried for a typical muffle furnace at the Firozabad glass industry, by the Tata Energy Research Institute reveals the following energy balance:

Flue gas losses	=	47.8%
Structural losses	=	36.0%
Heat loss due to CO formation	=	9.9%
Heat loss to trays	=	3.7%
Heat loss to bottom ash	=	1.1%
Heat gain by bangles	=	1.5%

The above result implies that the efficiency of the muffle furnace is as low as 2%. The existence of the scope for a considerable improvement, via the design of an energy efficient muffle furnace, and hence deriving the maximum economic benefits, has inspired the project undertaken.

2. BACKGROUND

2.1 HEAT DISTRIBUTION IN A FURNACE

Thermal efficiency of a furnace which is the ratio of the quantity of heat utilised to the quantity of heat supplied, varies in a wide range from as low as of 10% in some furnaces to 60% or even more than 90% in case of a well designed one [1].

The fuel required for combustion is burnt in the combustion zone of the furnace. Contrary to the requirements, not all the heat is absorbed by the stock. Some of it passes into the furnace wall and hearth. Another portion of heat is lost to the surrounding by radiation and convection from the outer surface of the walls or by conduction, into the ground. Heat is also radiated through cracks or other openings and furnace gases pass out around the door, frequently burning in the open and carrying off the heat. Heat is also lost each time the door is opened. Finally, the bulk of heat loss passes out along with the flue gases either in the form of sensible heat or as incomplete combustion.

Various factors which affect the fuel economy and furnace efficiency adversely are:

- Improper design
- Incomplete combustion
- Improper heat distribution
- Operation at a temperature much higher than desired temperature.
- Heat losses from walls, openings, cracks.
- Improper control of furnace draught
- No waste heat recovery from the furnace flue gases.

The present work is primarily concerned with heat balances. Hence the various kinds of heat losses are considered. Heat losses can be classified into following four categories:

1. Heat loss from furnace walls - Heat is lost to the surroundings by radiation and convection from outer surface of furnace walls and by conduction into the ground.
2. Heat loss through openings and crack - There may be various openings in the furnace, for example,peep holes,inspection doors. Charging and discharging doors, which cause considerable heat loss.
3. Heat loss through waste gases - In combustion type furnaces two additional sources of heat loss occur. These losses are caused by the heat energy that the product of combustion take out of the furnace in the shape of unburnt fuel and in the form of sensible heat due to high temperature. The heat loss is commonly known as stack loss.
4. Special losses
    a) Heat losses due to part of stock projecting out of the furnace - Heat is lost due to dissipation of heat from metal portion projecting from the inside of the furnace into the air. Throughout the heating period, heat flows from the part that is in the furnace to the part that is outside, and a portion of this heat is dissipated from the latter part, by radiation and convection.
    b) Heat loss to tongs and charging machines - Cold tongs absorb heat while in the furnace. Although the flow of heat into tongs is less than the heat that is radiated through the open door, it is by no means inconsiderable.
    c) Losses to trays,conveyer chains, and rollers.

2.2 DETAILS OF FIROZABAD GLASS FACTORY

The Firozabad glass factory accounts for nearly seventy percent of the glass production in the small sector in India. The cluster is an agglomeration of small units engaged in the manufacture of hollow ware, headlight covers, beads, decorative items and bangles. Principally, three types of furnace are employed by the Firozabad industry - tank furnaces, pot furnaces, muffle furnace. The muffle furnace commonly known as 'pakai bhatti' is used for baking bangles to give them the required finish.

There are about 400 registered muffle furnace units in the cluster. These furnaces work round the clock and for nearly 350 days a year. Coal is the primary fuel for these furnaces although wood is also used as a supplementary fuel in some of the units. Due to the irregular supply and sometimes the non-availability of coal from government quota, coal is generally purchased from open market at prices in the range of Rs. 2500-3000 per tonne. The schematic diagram of a typical muffle furnace used in Firozabad is shown in Fig. 1. Around 14,4000 bangles per day are baked in a single furnace. The bangles are baked in trays through three sequential baking hearth. The temperature in these are around 400,605,785 degree celsius in the top, middle, and bottom hearth, respectively. Bangles arranged on a steel tray are first baked on the top hearth for 55-60 seconds (low temperature); on middle hearth for 55-70 seconds; and finally on the bottom hearth for 85-105 seconds (high temperature). There is no temperature measuring device and the actual baking time is decided by the operator based on his experience.

The muffle furnace operates on the natural draught. Fuel (coal/wood) is fed over the fixed grate provided at the bottom of the furnace. Hot gases pass through a zig-zag path before leaving the chimney. The flue gases flow upwards around the hearth finally leaving the furnace at around 880

degree celsius. Heat is transferred from the flue gases to the hearth by conduction. The present cost of construction of one muffle furnace is about rupees 7000/- **[2]**.

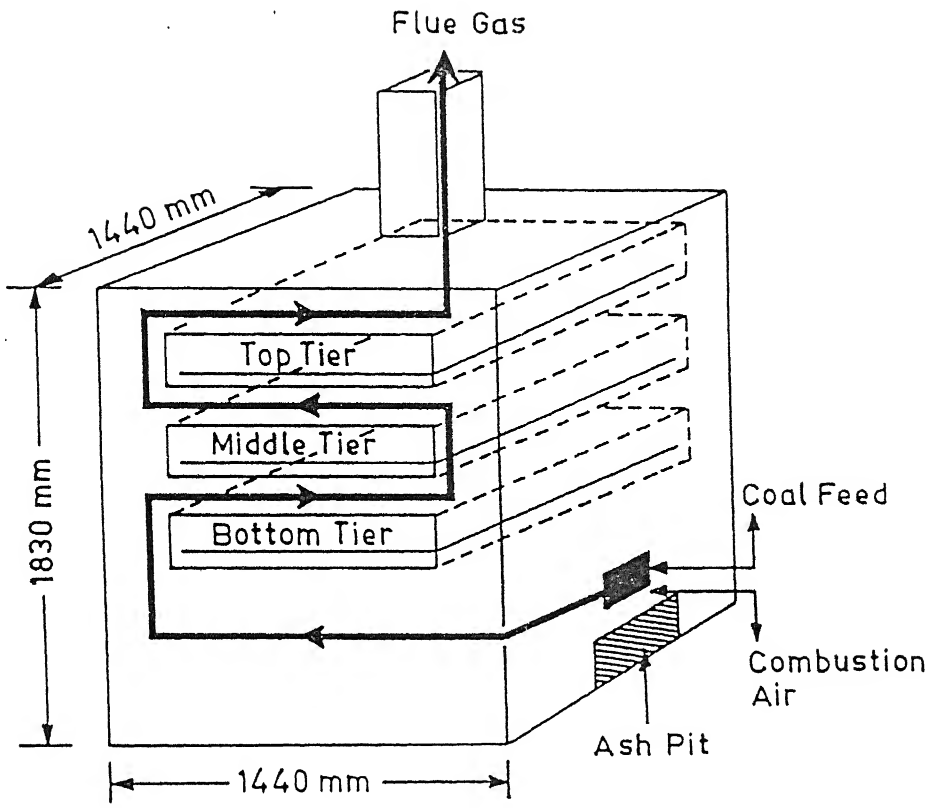

FIG. 1. SCHEMATIC DIAGRAM OF MUFFLE FURNACE.

3. PROBLEM FORMULATION

3.1 ASSUMPTIONS

The construction details of the furnace and the composition of the coal was not available. Hence the following assumptions have been made.

1) The coal used for the purpose is lignite. The composition (% weight) of which is :

   C = 59.9%

   H_2 = 4.37%

   O_2 = 18.64%

   N_2 = 1.22%

   S = 2.65%

   Ash = 13.22%

   The adiabatic flame temperature is 2060 degree celsius [3].

2) The actually obtained flame temperature is about 70% of the theoretical attainable temperature [4].

3) The coal particles are assumed to be perfect spheres with a diameter of about 40mm.

4) The coal pieces are so arranged that it forms a close packed bed.

5) The air enters the furnace at an ambient temperature (40 degree celsius).

6) The process is assumed to be in steady state.

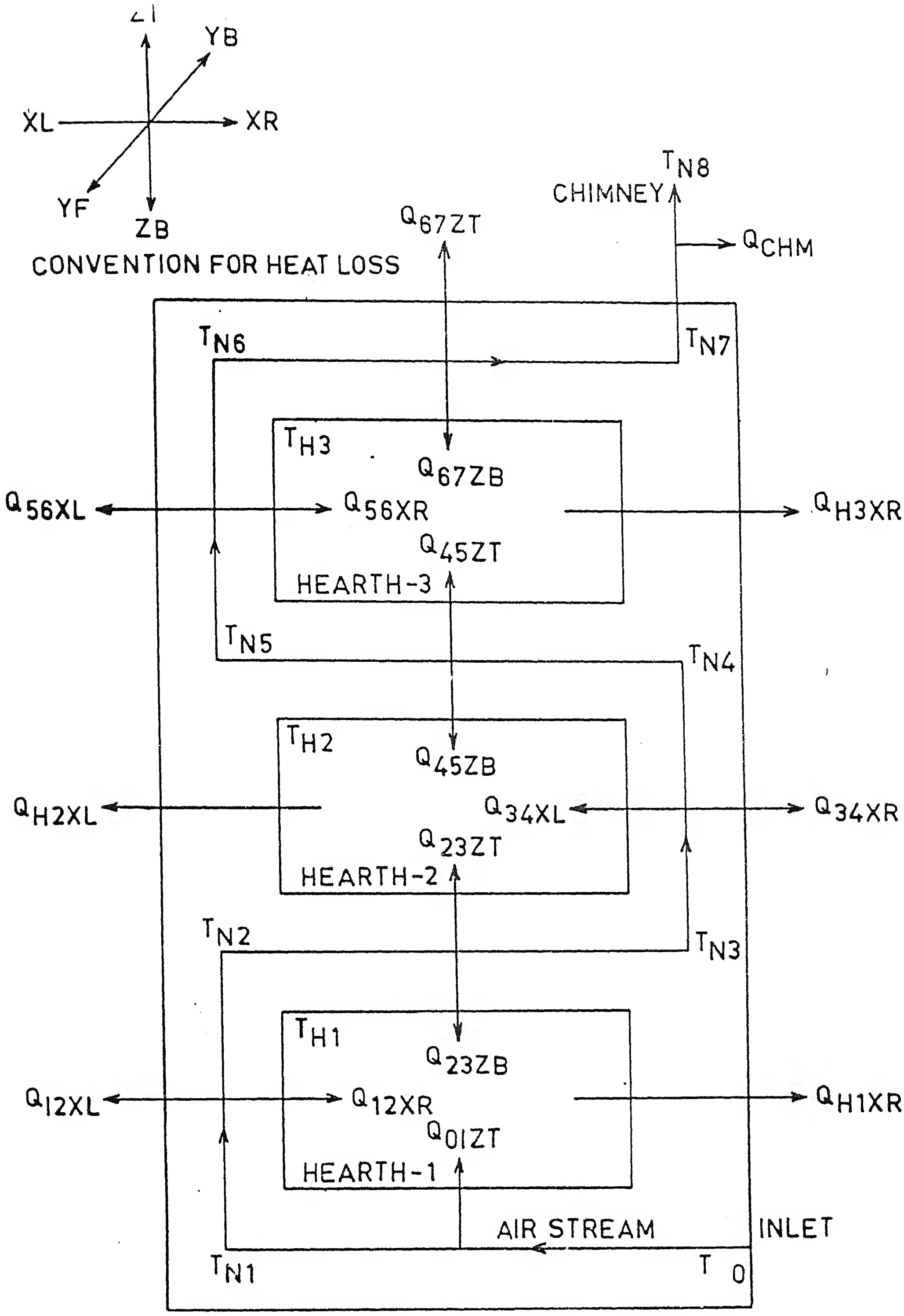

FIG. 2. SCHEMATIC DIAGRAM SHOWING HEAT LOSSES.

3.3 HEAT BALANCE EQUATIONS

The temperature should increase from T_0 to T_{N1} due to the net inflow of heat from combustion. Thereafter, it is expected to decrease because of losses to the hearth and the ambient.

The energy balance of the various air streams results in the following equations:

Stream 0-1

$$\dot{m}C_pT_i + \dot{Q}_{coal} = \dot{m}C_pT_{i+1} + Q_{O1ZT} + Q_{12YF} + Q_{12YB} + Q_{12XL}$$

$$\dot{m}C_pT_i + C_f vH_v(T_{coal}-T_a) = \dot{m}C_pT_{i+1} + \frac{[\frac{T_i+T_{i+1}}{2}-T_{HI}]}{R_1} + \frac{[\frac{T_i+T_{i+1}}{2}-T_a]}{R_2} + \frac{[\frac{T_i+T_{i+1}}{2}-T_a]}{R_3} + \frac{T_{i+1}-T_a}{R_4}$$

$$T_i(\dot{m}C_p-\frac{1}{2R_1}-\frac{1}{2R_3}-\frac{1}{2R_3}) - T_{i+1}(\dot{m}C_p+\frac{1}{2R_1}+\frac{1}{2R_2}+\frac{1}{2R_3}+\frac{1}{R_4})+\frac{T_{HI}}{R_1} =$$

$$-T_a(\frac{1}{R_2}+\frac{1}{R_3}+\frac{1}{R_4}) + C_fH_v(T_{coal}-T_a) \qquad (1)$$

where,

$$R_1 = \frac{NX}{A_{12ZT}}(\frac{D_{12ZT}}{k})$$

$$R_2 = \frac{NX}{A_{12YF}}(\frac{D_{12Yf}}{k}+\frac{1}{h_v})$$

$$R_3 = \frac{NX}{A_{12YB}}(\frac{D_{12YB}}{k}+\frac{1}{h_v})$$

$$R_4 = \frac{NX}{A_{12XL}}(\frac{D_{12XL}}{k}+\frac{1}{h_v})$$

Implicit in the equation 1 is the assumption that gas is in contact with coal along the entire length of stream 0,1. In the expression for Q_{coal}, an adjustable parameter, C_f, has been incorporated to account for the lack of information about (a) properties of the coal, (b) the nature of gas-solid contact on the grate and (c) heat losses to the ground and through furnace wall below the grate. R_1, R_2, R_3, R_4 represents the resistances (per unit interfacial area) to heat flow from the flue gas to different parts of furnaces, as shown in Fig. 2. Implicit in equation 1 is the asumption that the inner walls of the bottom most hearth is at a uniform temperature, T_{H1}. A simple energy balance on a differential element in stream 0,1 will show that the flue gas temperature varies exponentially. Hence the linear variation in flue gas temperature assumed in equation 1 can only be valid if each gas stream is divided in a number of small element (NX). Equation 1 has to be written for each element in stream 0,1. A similar methodology has been followed for carrying out energy balance for all other air stream shown in Fig. 2.

Stream 1-2

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1} + Q_{12XR} + Q_{12XL}$$

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1} + \frac{[\frac{(T_i+T_{i+1})}{2} - T_{HI}]}{R_5} + \frac{[\frac{T_i+T_{i+1}}{2} - T_a]}{R_6} \tag{2}$$

$$T_i(\dot{m}C_p - \frac{1}{2R_5} - \frac{1}{2R_6}) - T_{i+1}(\dot{m}C_p + \frac{1}{2R_5} + \frac{1}{2R_6}) + \frac{T_{HI}}{R_5} = -\frac{T_a}{R_6}$$

where,

$$R_5 = \frac{NZ}{A_{12XL}}(\frac{D_{12XL}}{k})$$
$$R_6 = \frac{NZ}{A_{12XR}}(\frac{D_{12XR}}{k})$$

Stream 2-3

$$\dot{m}c_pT_i = \dot{m}C_pT_{i+1}+Q_{23ZB}+Q_{23ZT}+Q_{23XL}+Q_{23XR}+Q_{23YF}+Q_{23YB}$$

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+\frac{\frac{(T_i+T_{i+1})}{2}-T_{H1}}{R_1}+\frac{\frac{(T_i+T_{i+1})}{2}-T_{H2}}{R_1}+(\frac{T_i-T_a}{R_4})+(\frac{T_{i+1}-T_a}{R_4})+\frac{(\frac{T_{i+1}+T_i}{2})-T_a}{R_7})+\frac{(\frac{T_{i+1}+T_i}{2})-T_a}{R_8}$$

$$T_i(\dot{m}C_p-\frac{1}{R_1}-\frac{1}{R_4}-\frac{1}{2R_7}-\frac{1}{2R_8})-T_{i+1}(\dot{m}C_p+\frac{1}{R_1}+\frac{1}{R_4}+\frac{1}{2R_7}+\frac{1}{2R_8})+\frac{T_{H1}}{R_1}+\frac{T_{H2}}{R_1} = -\frac{2T_a}{R_4}-\frac{T_a}{R_7}-\frac{T_a}{R_8}$$

(3)

where,

$$R_7 = \frac{NX}{A_{34YF}}(\frac{D_{34YF}}{k})$$
$$R_8 = \frac{NX}{A_{34YB}}(\frac{D_{34YB}}{k})$$

Stream 3-4

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+Q_{34XL}+Q_{34XR}$$

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+\frac{\frac{(T_{i+T_i+1})}{2}-T_{H2}}{R_5}+\frac{\frac{(T_i+T_{i+1})}{2}-T_a}{R_6} \tag{4}$$

$$T_i(\dot{m}C_p-\frac{1}{2R_5}-\frac{1}{2R_6})-T_{i+1}(\dot{m}C_p+\frac{1}{2R_5}+\frac{1}{2R_6})+\frac{T_{H2}}{R_5} = -\frac{T_a}{R_6}$$

Stream 4-5

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+Q_{45ZB}+Q_{45ZT}+Q_{45XL}+Q_{45XR}+Q_{45XF}+Q_{45YB}$$

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+\frac{(\frac{T_i+T_{i+1}}{2})-T_{H2}}{R_1}+\frac{(\frac{T_i+T_{i+1}}{2})-T_{H3}}{R_1}+\frac{T_{i+1}-T_a}{R_4}+\frac{T_{i+1}-T_a}{R_4}+\frac{(\frac{T_i+T_{i+1}}{2})-T_a}{R_7}+\frac{(\frac{T_i+T_{i+1}}{2})-T_a}{R_8}$$

$$T_i(\dot{m}C_p-\frac{1}{R_1}-\frac{1}{2R_7}-\frac{1}{2R_8})-T_{i+1}(\dot{m}C_p+\frac{1}{R_1}+\frac{3}{R_4}+\frac{1}{2R_7}+\frac{1}{2R_8})+\frac{T_{H2}}{R_1}+\frac{T_{H3}}{R_1} = -T_a(\frac{2}{R_4}+\frac{1}{R_7}+\frac{1}{R_8}) \tag{5}$$

Stream 5-6

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+Q_{56XR}+Q_{56XL}$$

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+(\frac{\frac{(T_i+T_{i+1})}{2}-T_{H3}}{R_5})+(\frac{\frac{(T_i+T_{i+1})}{2}-T_a}{R_6}) \tag{6}$$

$$T_i(\dot{m}C_p-\frac{1}{2R_5}-\frac{1}{2R_6})-T_{i+1}(\dot{m}C_p+\frac{1}{2R_5}+\frac{1}{2R_6})+\frac{T_{H3}}{R_5} = -\frac{T_a}{R_6}$$

Stream 6-7

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+Q_{67ZB}+Q_{67ZT}+Q_{67XL}+Q_{67XR}+Q_{67YF}+Q_{67YB}$$

$$\dot{m}C_pT_i - \dot{m}C_pT_{i+1} = \frac{(\frac{T_i+T_{i+1}}{2})-T_{H3}}{R_1}+\frac{(\frac{T_i+T_{i+1}}{2})-T_a}{R_9})+\frac{(T_i+T_a)}{R_4}+\frac{(T_{i+1}-T_a)}{R_7}+\frac{(\frac{T_i+T_{i+1}}{2})-T_a}{R_7}+\frac{(\frac{T_i+T_{i+1}}{2})-T_a}{R_8}$$

$$T_i(\dot{m}C_p-\frac{1}{2R_1}-\frac{1}{R_4}-\frac{1}{2R_7}-\frac{1}{2R_8}-\frac{1}{2R_9})-T_{i+1}(\dot{m}C_p+\frac{1}{2R_1}+\frac{1}{2R_4}+\frac{1}{2R_7}+\frac{1}{2R_8}+\frac{1}{2R_9})+\frac{T_{H3}}{R_1}$$
$$=-T_a(\frac{2}{R_4}+\frac{1}{R_7}+\frac{1}{R_8}+\frac{1}{R_9}) \qquad (7)$$

where,

$$R_9 = \frac{NX}{A_{67ZT}}(\frac{D_{67ZT}}{k}+\frac{1}{h_h})$$

Stream 7-8

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+Q_{chm}$$

$$\dot{m}C_pT_i = \dot{m}C_pT_{i+1}+(\frac{(\frac{T_i+T_{i+1}}{2})-T_a}{R_{10}}) \qquad (8)$$

$$T_i(\dot{m}C_p-\frac{1}{2R_{10}})-T_{i+1}(\dot{m}C_p+\frac{1}{2R_{10}}) = -\frac{T_a}{R_{10}}$$

where,

$$R_{10} = \frac{NC}{A_{chm}}\left(\frac{1}{h_v} + \frac{D_{chm}}{k}\right)$$

Three more equation can be obtained by doing a steady state heat balance on each of the three hearths:

Hearth - 1

$$Q_{01ZT} + Q_{12XR} + Q_{23ZB} = Q_{H1XR} + Q_{H1YF} + Q_{H1YB}$$

where,

$$Q_{H1XR} = \frac{T_{H1} - T_a}{R_{11}}$$

$$Q_{H1XR} = \frac{T_{H1} - T_a}{R_{12}}$$

$$Q_{H1XR} = \frac{T_{H1} - T_a}{R_{13}}$$

$$R_{11} = \frac{D_{H1XR}}{k} + \frac{1}{h_v}$$

$$R_{12} = \frac{D_{H1YF}}{k} + \frac{1}{h_v}$$

$$R_{13} = \frac{D_{H1YB}}{k} + \frac{1}{h_v}$$

Hearth - 2

$$Q_{23ZT} + Q_{34XR} + Q_{45ZB} = Q_{H2XR} + Q_{H2YF} + Q_{H2YB}$$

where,

$$Q_{H2XR} = \frac{T_{H2} - T_a}{R_{11}}$$

$$Q_{H2YF} = \frac{T_{H2} - T_a}{R_{12}}$$

$$Q_{H2YB} = \frac{T_{H2} - T_a}{R_{13}}$$

Hearth - 3

$$Q_{45ZT} + Q_{56XR} + Q_{67ZB} = Q_{H3XR} + Q_{H3YF} + Q_{H3YB}$$

where,

$$Q_{H3XR} = \frac{T_{H3} - T_a}{R_{11}}$$

$$Q_{H3YF} = \frac{T_{H3} - T_a}{R_{12}}$$

$$Q_{H3YB} = \frac{T_{H3} - T_a}{R_{13}}$$

Equation of the form 1 to 11 are then solved simultaneously after dividing the stream into an "adequate" number of subsections.

4. CORRELATIONS

The various correlations used for the calculation purposes discussed below. The solution of equation 1 to 11 requires a prior knowledge of the gas mass flow rate. It can be obtained with the help of Bernoulli's equation, as shown below.

4.1) Bernoulli's equation for the calculation of mass flow rate [5]:

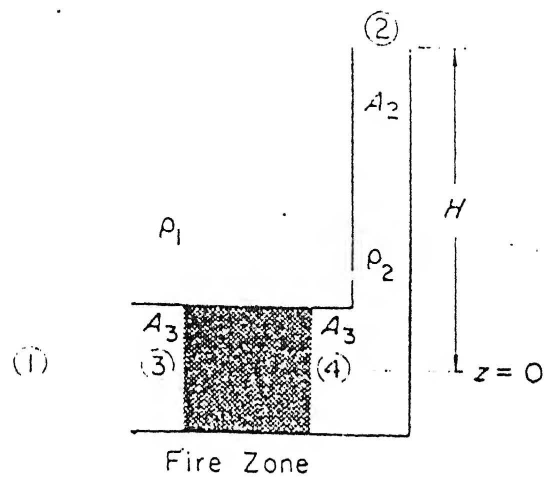

Fig. 5 Natural convection in a chimney.

As air heats up in a fire place it expands so that the gas flowing inside the chimney stack is lighter than the air outside. The pressure at the lower and upper ends of the stack are determined by the relatively denser outside air, so that

$$(P_2-P_1)= - \rho_1 \, g \, H$$

where,

H = height of the chimney stack

ρ_1 = density of the cooler air outside

ρ_2 = density of the warmer air inside

The lighter air inside the chimney stack, therefore experiences buoyancy and moves upwards. Since the density of fluid is not constant everywhere, the mechanical energy equation cannot be applied between regions 1 and 2. The problem is simplified, if the flow is divided into three regions. The first region extends from 1 to 3 where the colder air flows at constant density ρ_1 from the relatively stagnant region into the fire zone of cross sectional area A_3. The second region is the fire zone itself where heat is added to the air such that its temperature rises at essentially constant pressure. In the third region from section 4 to 2 the lighter gas flows (at constant density ρ_2) into the stack of area A_2 finally exiting into the atmosphere at pressure P_2. The mechanical energy equation applied to the first and third region to give (on neglecting losses):

$$\frac{P_1}{\rho_1} = \frac{1}{2}V_3^2 + \frac{P_3}{\rho_1}$$

and

$$\frac{V_4^2}{2} + \frac{P_4}{2} = \frac{V_2^2}{2} + \frac{P_2}{\rho_2} + gH$$

The continuity equation relates the velocities at sections 2, 3 and 4 as

$$\rho_1 V_3 A_3 = \rho_2 V_4 A_3 = \rho_2 V_2 A_2$$

Expressing all the velocities in terms of the desired stack velocity V_2, also, $P_3 = P_4$ and

$$P_2 - P_1 = -\rho_1 g H$$

we get

$$V_2 = \sqrt{\frac{2(\frac{\rho_1}{\rho_2}-1)gH}{(1-(\frac{A_2}{A_1})^2)(1-\frac{\rho_2}{\rho_1})}}$$

The mass flow rate is calculated as shown:

$$\dot{m} = \rho_2 A_2 V_2$$

4.2) Calculation of heat transfer coefficient for the packed coal bed

The coal is burnt below the bottommost hearth. As assumed, the perfect spherical coal particles form a packed bed. The heat transfer coefficient for the same is given by [6]:

$$h_v = 12xv_{og}x(\frac{t^{0.3}}{d_p^{1.35}})$$

where,

h_v = hs = volumetric heat transfer coefficient, kcal/cubic metre-hr degree celsius.

t = temperature in degree celsius.

d_p = particle diameter, mm.

vog = superficial gas velocity, m/sec

4.3) Calculation Of Heat Transfer Coefficient across vertical plates

For the free convection across the vertical plane surfaces, the correlation for the plate length Nusselt number is given by

$$Nu_l = CRa_l^m$$

where the coefficient C and the exponent m are functions of Ra_l, the rayleigh number based on the total plate length. In laminar region ($10^4 < Ra_l < 10^9$), m = 1/4. For the turbulent region ($10^9 < Ra_l$) m = 1/3.

The correlation that has been used is [7]:

$$Nu_l = 0.68 + 0.670 (Ra_l^{0.25}) [1 + ((0.492/Pr)^{9/16})]^{(-4/9)}$$

for $0 < Ra_l < 10^9$

$$Nu_l = \{0.825 + 0.387 (Ra_l^{1/6}) [1 + ((0.492/Pr)^{9/16})]^{(-8/27)}\}^2$$

for $10^9 < Ra_l$

5. RESULT & DISCUSSION

ALGORITHM

1) Assume values for CF, XN, ZN and CN

2) Guess the value for T_{N8} , temperature of the flue gas coming from the stack.

3) Calculate mdot using Bernoulli's equation.

4) Calculate temperature profile in the furnace by solving equations 1 to 11 for each element in the flue gas stream.

5) If convergence in temperature profile not achieved then go to step (3) using the new value of T_{N8}.

Calculation is carried out for the furnace depicted in fig. 1. The internal dimensions of the furnace such as (a) dimensions of the flow channel, (b) hearth dimensions and (c) wall thickness are not known. Some of these information have been estimated from fig. 1. by assuming that the furnace has been drawn to scale.

Initially calculations were carried out by dividing each section of the flow stream into 10 sections(XN=ZN=CN=10).The ambient air ,and hence T is assumed to be 40 degree Celsius. The value of CF was varied until the value of T_{N8} =850 degree Celsius was obtained. It may be recalled that this is the measured flue gas exit temperature. Table 1 shows the temperature distribution inside the furnace at CF=0.04 when TN8=850 degree Celsius is achieved.

Table 1 : Temperature profile at CF = 0.04, and XN = ZN = CN = 10

Sr. No.	Symbol Used	Temperature in ^0C
1	T_0	5874.31
2	T_{N1}	3073.37
3	T_{N2}	2492.41
4	T_{N3}	1660.06
5	T_{N4}	1855.25
6	T_{N5}	1768.20
7	T_{N6}	1608.10
8	T_{N7}	857.31
9	T_{H1}	3281.14
10	T_{H2}	1811.58
11	T_{H3}	1978.79

MASS FLOW $= 2.1736 * 10^{-3}$

Increasing the number of division in each stream to 50 gives the following results

Table 2: Temperature profile at CF = 0.04, XN = ZN = CN =50

Sr. No.	Symbol Used	Temperature in ^0C
1	T_0	5745.42
2	T_{N1}	3254.61
3	T_{N2}	2911.06
4	T_{N3}	2148.68
5	T_{N4}	2656.78
6	T_{N5}	2709.27
7	T_{N6}	2470.20
8	T_{N7}	1172.88
9	T_{H1}	3564.43
10	T_{H2}	2384.02
11	T_{H3}	3044.68

MASS FLOW = $1.855 * 10^{-3}$

Using XN = ZN= CN = 200, also does not significantly alter the nature of results shown in table 1 and Table II clearly using T_{N8} has a convergence criteria is not desirable because it lead to unusually high value temperature. The measured hearth temperature were 400, 605, 785^{0}C in the top, middle, and bottom hearth respectively. Moreover it is also evident that the predicted temperature profile does not decrease continuously as expected and let the temperature of the topmost hearth is greater then the metal hearth. The second approach for solving this problem is to use a value of CF such that the calculated hearth temperature are similar to the measured values. Table III and IV depict the temperature profile for CF = 0.007 when each stream is divided into 10 and 50 section respectively.

Table 3 : Temperature profile at CF = 0.007, and XN = ZN = CN = 10

Sr. No.	Symbol Used	Temperature in ^0C
1	T_0	1374.80
2	T_{N1}	763.975
3	T_{N2}	603.265
4	T_{N3}	434.578
5	T_{N4}	488.373
6	T_{N5}	481.532
7	T_{N6}	440.968
8	T_{N7}	285.885
9	T_{H1}	770.594
10	T_{H2}	460.722
11	T_{H3}	535.754

MASS FLOW $= 2.895 * 10^{-3}$

Table 4: Temperature profile at CF = 0.007, XN = ZN = CN =50

Sr. No.	Symbol Used	Temperature in ^0C
1	T_0	1473.06
2	T_{N1}	878.159
3	T_{N2}	764.323
4	T_{N3}	598.351
5	T_{N4}	737.095
6	T_{N5}	763.069
7	T_{N6}	700.382
8	T_{N7}	432.864
9	T_{H1}	912.145
10	T_{H2}	648.375
11	T_{H3}	856.491

MASS FLOW = $2.726 * 10^{-3}$

Even though the temperature profile of the flow stream for XN = ZN = CN = 50 seems to be more realistic, the middle and top hearth temperature are inconsistent with observations.

To get realistic value of temperatures, the procedure for solving the hearth temperature was modified. Instead of solving the three hearth temperatures T_{H1}, T_{H2} & T_{H3}, using the energy valance approach as manifested in equations 9 - 11, T_{H1} to TH3 was calculated as follows :

$$T_{H1} = 0.1\ (_T + T_{N1} + T_{N2} + T_{N3})$$

$$T_{H2} = 0.1\ (T_{N2} + T_{N3} + T_{N4} + T_{N5})$$

$$T_{H3} = 0.1\ (T_{N4} + T_{N5} + T_{N6} + T_{N7})$$

A coefficient of 0.1 was used in the equation instead of 0.1 to ensure the that the hearth temperature was lower than that of the gas stream flowing around it. Unfortunately, this scheme also did not provide the desired results.

The main reason for improper prediction of temperature seems to be the low mass flow rate of flue gas inside the furnace. Lower mass flow rates imply greater heat dissipation to the surrounding and hence a greater drop in temperature of the flow stream, a fact that can be easily verified the energy balance equations listed in the previous chapter. One method of overcoming w in problem is to divide the stream into greater number of division. Unfortunately, dividing each stream into as many as 200 section, that is dividing the entire flow path into the furnace into 1600 section did not rectified the situation clearly the problem lies elsewhere.

It is seen that the most plausible cause for the factors to predict the temperature profile in the muffle furnace is the assumption that all the hearth walls are at a uniform temperature. At low mass flow rate the temperature of the flue gas would be expected to be in equilibrium with the hearth walls . That is the temperature of the hearth wall should very enough manner similar to the flue gas stream adjacent to it, although its temperature would be lower than the flue gas temperature. For example the temperature of the hearth will just above point) would be the highest and it would decrease continuously upto N3. A similar situation should exist in the other two hearths.

Therefore assumption of uniform hot temperature resulted in larger heat loses from the flow stream. Hence we have a situation where heat flows from, example a hearth 1 to stream 2 - 3 (b) hearth 2 to stream 4 - 5 and (C) hearth 3 to stream 7 - 8.

Future Work

The energy balance equations stated in the previous chapter has to be modified to account for the non-uniform hearth temperature.

(1) Equation 1 to 8 should be modified & make temperature T_{H1} to T_{H3} as variables

(2) Replacing equation 9 to 11 with "radiation's" equation.

CONCLUSIONS

The main reason for improper prediction of temperature seems to be the low mass flow rate of flue gas inside the furnace. Lower mass flow rates imply greater heat dissipation to the surrounding and hence a greater drop in temperature of the flow stream , a fact that can be easily verified by the energy balance equations listed in the previous chapter. One method of overcoming this problem is to divide the stream into greater number of divisions. Unfortunately, dividing each stream into as many as 200 sections, that is dividing the entire flow path in the furnace into 1600 section did not rectify the situation. Clearly the problem lies elsewhere.

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Therefor assuption of uniform hearth temperature resulted in larger heat losses from the flow stream. Hence we have a situation where heat flows from, example a hearth 1 to stream 2-3 (b). hearth 2 to stream 4-5 and (c) hearth 3 to stream 7-8.

FUTURE WORK

The energy balance equations stated in the previous chapter has to be modified to account for the non-uniform hearth temperature.

(1). Equation 1 to 8 should be modified & make temperature T_{H1} to T_{H3} as variables.

(2). Replacing equation 9 to 11 with "radiation's" equation.

(3). The problem should be worked under steady state conditions.

(4). The temperature change should be two-dimensional. So differential equations have to be solved.

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